

Semi-analytical modelling of a hydraulic free-piston engine



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HIGHLIGHTS

- A semi-analytical method for the HPFE parameter design is proposed.
- The HPFE piston oscillation is a hard self-oscillation.
- The piston oscillation has an unsymmetrical oscillation characteristic.
- Stable running of a single piston HPFE prototype is realised.

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ABSTRACT

The hydraulic free-piston engine is a hybrid of a hydraulic plunger pump and a free-piston engine. This paper presents a semi-analytical method for the hydraulic free-piston engine parameter design. An oscillation function described the piston oscillation were established. The piston oscillation characteristics were analysed. The parameter sensitivity and the limit cycle stability of the piston oscillation were investigated. The results show that the single piston free-piston engine system is a hard self-oscillation with variable damping and stiffness. External energy is required to start the piston oscillation. The piston oscillation has an unsymmetrical oscillation characteristic. The dead centres of the piston motion are determined by the energy balance. The energy supply and the energy consumption are determined by the generalised damping in the piston oscillation function. The piston oscillation frequency should satisfy the minimum compression ratio requirement for the compression ignition. In the steady operation, the piston motion is a self-excitation oscillation. The trajectory stability is mainly determined by the fuel combustion energy supply. The scavenging affected by the BDC should be paid special attention to ensure the trajectory stability.

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1. Introduction

The free-piston concept was firstly proposed by Pescara in 1920s [1] and developed fast from 1930s to 1960s [2,3]. However, for many reasons, the free-piston engine (FPE) was replaced by advanced crank internal-combustion engines and practically abandoned since the 1960s [3]. Potential advantages of the FPE, such as optimised combustion through variable compression ratio, and lower frictional losses due to the simple structure with few moving parts, makes the FPE concept receive more interest in recent years [4–7].

Compared to the conventional crank internal-combustion engine, the FPE has two typical features [2,3]:

- (a) The piston motion is not restricted by a rotating crankshaft, as known from crank engines.

- (b) The power is taken out by media, not by gear or shaft.

The FPE concept can be divided into three categories according to the number or position of the piston. With different media of loads, the FPE can be converted into different power engines. Most of them are designed as power sources for automotive applications [3]. The General Motors XP-500 completed in 1957 was the first free-piston automobile [3]. The XP-500 was powered by the General Motors Research 4-4 ‘Hyprex’ engine. The General Motors Research 4-4 ‘Hyprex’ was a dual opposed piston, diesel free-piston engine. It had a power output of around 185 kW [8]. Because of technical shortcomings and more promising research in other areas of alternative propulsion, General Motors halted further development on the free-piston automobile within three years of the XP-500’s completion [9]. Ford, like General Motors, saw a potential in the free-piston gas generator for automotive applications in the mid-1950s [3]. Frey and co-researchers proposed an analytical model for the design of a free-piston gas generator and described the development of a 112 kW prototype [10]. The

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Nomenclature

a	amplitude	p_{gC}	gas pressure brought by the fuel combustion
A_1	function of the amplitude	p_{gV}	gas pressure in the pure compression process
b	decentration	p_w	hydraulic working pressure
c	generalised damping	$Para$	design parameter value
c_{equi}	equivalence damping	$Para_{rated}$	design parameter rated value
c_o	damping coefficient	S	piston cross-section area
D	cylinder bore	S_1	cross-section area of the check chamber
E_{cyc}	energy released by fuel combustion in one cycle	S_2	cross-section area of the pump chamber
f	piston oscillation frequency	S_3	cross-section area of the compression chamber
f_c	simplified combustion gas force	V_0	initial value of the in-cylinder gas volume
F_{ch}	check chamber force	y	state variable
F_{co}	compression chamber force	x	piston displacement
F_g	in-cylinder gas force	x^*	displacement in the calculation coordinate system
F_{gC}	force brought by the fuel combustion	x_1^*	function of the amplitude
F_{gCmax}	maximum value of force brought by the fuel combustion	x_{CSBDC}	BDC in the compression stroke
F_{gV}	force determined by the changing of gas volume	x_{ESBDC}	BDC in the expansion stroke
F_{pu}	pump chamber force		
F_r	friction force		
F_{r0}	value of friction force		
K	generalised nonlinear stiffness		
L	distance between the piston balance position and the cylinder head plate		
m	moving piston mass		
p	in-cylinder gas pressure		
p_0	initial value of the in-cylinder gas pressure		
p_c	hydraulic compression pressure		

Greek symbols

γ	in-cylinder gas adiabatic exponent
δ	active range of the combustion force
ε	compression ratio
η_i	indicated thermal efficiency
φ	phase
Φ_0 and Φ_1	functions of the amplitude and the phase
ω	change rate

favourable torque-speed characteristic of the prototype led Ford engineers to conclude that the Ford model 519 free-piston power plant was ideally suited for installation in the Ford Typhoon tractor [3]. A hydraulic free-piston engine (HFPE) has been proposed as a low-cost and high-efficiency power unit for a fork-lift truck by Innas [4]. The HFPE is a combination of a free-piston engine and a hydraulic pump. The HFPE gets higher efficiency compared to the conventional assemble of a crank internal-combustion engine and a hydraulic plunger pump [11].

The free-piston engine characteristic is often investigated by discrete numerical methods [12–22,7]. Zero-dimensional modelling of free-piston engines has been proposed [12–19]. Some have investigated free-piston engines using multidimensional simulation models [20–22,7]. The numerical methods could achieve the instantaneous displacement, velocity and acceleration of the engine piston with a high degree of precision. However, the numerical result only contains the discrete numerical solution and the panorama of the system solution is still not completely offered. To achieve high precision numerical result, the numerical model is always complicated and the time-consuming for the model calculation is long. The numerical result is inconvenient for analysing the piston oscillation and the contributing factors. The result also cannot fully satisfy the requirement for the engine parameter control system design. A semi-analytical model has been developed for the free-piston Stirling engine analysis [23].

This paper studies the design method of a single piston hydraulic free-piston engine (SPHFPE) as the power source of a hydraulic hybrid vehicle. Different from the discrete numerical method used in the engine parameter design, a semi-analytical method was used. The forces applied on the piston were analysed and the piston oscillation semi-analytical model was established. Extensive results of the oscillation model were presented, giving insight into piston oscillation characteristics. The transient process between different stable piston trajectories was analysed. The results

indicate that the steady state of the designed free-piston engine is a hard self-excited vibration with variable damping and stiffness. The stable piston trajectories only appear in a special piston stroke range, which means that the engine is operated in energy balance conditions. The aim is to offer the piston oscillation panorama by the semi-analytical method.

2. Configuration of the SPHFPE

The configuration of the SPHFPE is shown in Fig. 1, firstly proposed by Innas BV [4]. The SPHFPE is a two-stroke diesel engine with uniflow scavenging and direct fuel injection. The exhaust valves and the scavenging pump are driven by the high-pressure

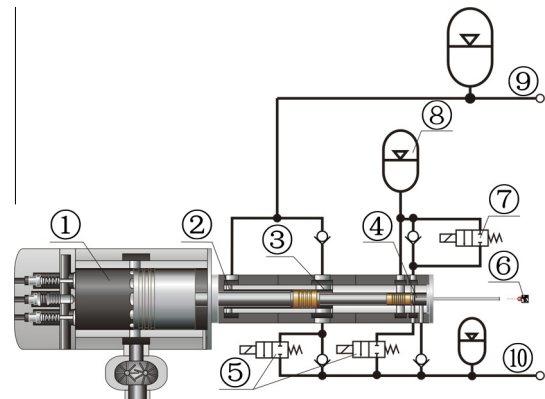


Fig. 1. SPHFPE configuration. ① Combustion cylinder, ② check chamber, ③ pump chamber, ④ compression chamber, ⑤ reset valve, ⑥ piston displacement transducer, ⑦ frequency control valve, ⑧ compression accumulator, ⑨ high-pressure rail, ⑩ low-pressure rail.

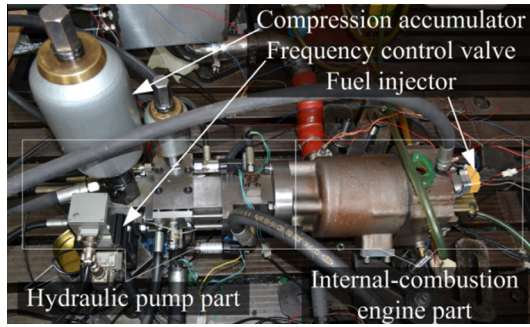


Fig. 2. SPHFPE prototype.

rail. For the injection system, a hydraulic-electronic unit injector from Caterpillar is used, as shown in Fig. 2.

The hydraulic pump of the SPHFPE consists of the check chamber, the pump chamber and the compression chamber. The check chamber restricts the rebound of the piston around bottom dead centre (BDC) after the expansion stroke. The pump chamber delivers one part of the high pressure oil in the expansion stroke. The other is delivered by the check chamber in the compression stroke. The compression chamber completes the compression stroke by the hydraulic power stored in the compression accumulator. The frequency control valve is used to control the working frequency of the SPHFPE. The design parameters of the SPHFPE prototype are:

- Piston diameter: 98.5 mm.
- Stroke: 114–125 mm.
- Mass of the moving piston: 4.2 kg.
- Pressure of the high-pressure rail: 20–30 MPa.
- Pressure of the low-pressure rail: 1.0–2.0 MPa.
- Hydraulic compression pressure: 15–16 MPa.
- Maximum output power: 15 kW.

3. Forces on the piston

The forces on the piston include in-cylinder gas force, hydraulic forces and friction forces, as shown in Fig. 3. The in-cylinder gas force F_g is brought by the in-cylinder gas pressure p . The force can be divided into two components. The force F_{gV} is determined by the changing of gas volume. The force F_{gC} is brought by the fuel combustion. The relation between the two gas forces is

$$F_g = F_{gV} + F_{gC} \quad (1)$$

The system coordinate was chosen with the origin at the cylinder head plate and the positive direction points to BDC. A dynamic model of the piston motion was developed based on Newton's second law:

$$m\ddot{x} = F_g + F_{ch} - F_{pu} - F_{co} - F_r \text{sign}(\dot{x}) - c_o \dot{x} \quad (2)$$

where m is moving piston mass, x is piston displacement, $F_r \text{sign}(\dot{x})$ is friction force, c_o is constant damping coefficient.

Substituting (1) into (2), we can get

$$m\ddot{x} = F_{gV} + F_{gC} + F_{ch} - F_{pu} - F_{co} - F_r \text{sign}(\dot{x}) - c_o \dot{x} \quad (3)$$

Eq. (3) can be rewritten as:

$$m\ddot{x} + c(\dot{x})\dot{x} + K(x, \dot{x})x = 0 \quad (4)$$

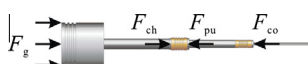


Fig. 3. Forces applied to the piston.

where

$$c(\dot{x}) = \frac{F_{pu} + F_r \text{sign}(\dot{x}) - F_{gC}}{\dot{x}} + c_o \quad (5)$$

$$K(x, \dot{x}) = \frac{F_{ch} + F_{co} - F_{gV}}{x} \quad (6)$$

Eqs. (3)–(5) describe a single degree freedom nonlinear vibration system. The $c(\dot{x})$ represents a generalised damping, determined by the friction force, the pump chamber force, the combustion gas force and the piston velocity. The $K(x, \dot{x})$ represents a generalised nonlinear stiffness, determined by the check chamber force, the compression chamber force, the volume changing gas force and the piston displacement. The generalised nonlinear stiffness does not consume or supply energy in one engine cycle.

The signs of the generalised damping items varying with the piston displacement are shown in Fig. 4. The plus sign means absorbing the piston kinetic energy, while the minus sign means releasing. The piston displacement is asymmetric according to top dead centre (TDC) [24]. The friction force part of the generalised damping is always to be positive in engine cycles, which means that it consumes the piston kinetic energy all the time. The sign of the pump chamber force changes to be the plus sign when the piston crosses TDC. This indicates that the piston kinetic energy is absorbed by the pump chamber force in the expansion stroke. The combustion gas force appears only after the injection of fuel. The sign of the combustion gas force part changes to be negative. It means that the energy is supplied by the fuel combustion and the piston kinetic energy is increasing. Further, if the combustion gas force appears in the compression stroke, it results in a decrease of the compression ratio.

To make the piston moving continually, the energy consumed and supplied by the generalised damping should be equal and enough. Based on the piston motion characteristics, the piston oscillation is a self-oscillation. When the energy supplied by the combustion gas force increases, the piston stroke also rises cycle by cycle. With the increase of the piston stroke, the energy consumed by the friction force and the pump chamber force also increases until the energy balance is achieved again. When the releasing energy decreases, the piston stroke falls until the energy balance is achieved in the same way. Namely, if the energy supplied by the combustion gas force is provided stably cycle by cycle, the piston reciprocating motion continues. If the supplied energy keeps being constant, the piston stroke is also stable and the piston oscillation becomes steady. The piston steady oscillation is a self-excited vibration. The steady phase trajectory of the piston is a limit cycle, which is a closed trajectory in the phase space. For the nonlinear vibration of the piston, the limit cycle means that at least one other trajectory of the piston spirals into it as time approaches infinity. The piston stroke becomes stable and the energy balance is achieved at last.

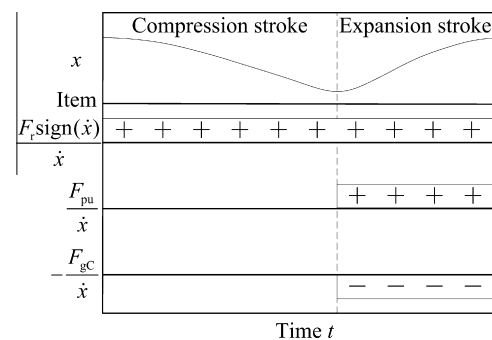


Fig. 4. Signs of the generalised damping items.

4. Piston oscillation model

The piston oscillation characteristics are analysed by the phase plane method:

$$\begin{cases} \dot{y} = \dot{x} \\ \dot{\dot{y}} = \frac{c(\dot{x})\dot{x} + K(x)\dot{x}}{m} \end{cases} \quad (7)$$

where y is state variable.

For further understanding of the system, the analytical method is used to get the piston limit cycle. The check chamber force and the compression chamber force are assumed to be constant. The pump chamber force is only applied on the piston in the expansion stroke. Eqs. (8)–(10) give the expressions of the forces:

$$F_{ch} = p_w S_1 \quad (8)$$

$$F_{co} = p_c S_3 \quad (9)$$

$$F_{pu} = \frac{p_w S_2 \text{sign}(\dot{x}) + p_w S_2}{2} \quad (10)$$

where p_w is hydraulic working pressure, S_1 is cross-section area of the check chamber, p_c is hydraulic compression pressure, S_3 is cross-section area of the compression chamber, S_2 is cross-section area of the pump chamber.

The volume changing gas force is calculated by

$$F_{gv} = p_0 \left(\frac{V_0}{V} \right)^\gamma S \quad (11)$$

where p_0 is initial value of the in-cylinder gas pressure, V_0 is initial value of the in-cylinder gas volume, S is piston cross-section area, γ is in-cylinder gas adiabatic exponent.

As shown by Fig. 5(a), p_{gv} is in-cylinder gas pressure in the pure compression process and p_{gc} is gas pressure brought by the fuel combustion. In order to simplify the complex fuel combustion process, the force applied by fuel combustion is replaced by the force shown by Fig. 5(b). The force is also assumed to actuate only in the expansion stroke. The simplified combustion force is easily described by a simple mathematical expression and very useful for

raising the model simulation speed. It seems that proper precision is also achieved, as shown in Fig. 5. The simplified combustion gas force can be expressed as:

$$F_{gc} = f_c(x, \dot{x}) \quad (12)$$

Based on the energy conservation law, the force maximum value $F_{gc\max}$ should satisfy:

$$E_{cyc} \eta_i = \frac{F_{gc\max} \delta}{2} \quad (13)$$

where E_{cyc} is energy released by fuel combustion in one cycle, η_i is indicated thermal efficiency, δ is action length of the force F_{gc} .

The friction force is

$$F_r \text{sign}(\dot{x}) = F_{r0} \text{sign}(\dot{x}) \quad (14)$$

where F_{r0} is value of the friction force. The value is a constant.

To get the limit cycles, a system of coordinates was chosen having the origin at the piston moving balance position and the positive direction pointing to TDC. Then Eq. (3) can be transformed to:

$$m\ddot{x}^* + \frac{\pi D^2 p_0}{4} \left[1 + \gamma \left(\frac{x^*}{L} \right) + \frac{\gamma(\gamma+1)}{2!} \left(\frac{x^*}{L} \right)^2 + \frac{\gamma(\gamma+1)(\gamma+2)}{3!} \left(\frac{x^*}{L} \right)^3 \dots \right] + f_c(x^*, \dot{x}^*) + p_w \left(S_1 - \frac{S_2}{2} \right) + \frac{(2F_r + p_w S_2) \text{sign}(\dot{x}^*)}{2} - p_c S_3 = 0 \quad (15)$$

where x^* is displacement in the calculation coordinate system, D is cylinder bore, L is distance between the piston balance position and the cylinder head plate.

With a simplified method, Eq. (15) is represented as

$$\ddot{x}^* + g(x^*) = \frac{1}{m} f(x^*, \dot{x}^*) \quad (16)$$

where

$$g(x^*) = \frac{p_0 \gamma \pi \left(\frac{D}{2} \right)^2}{mL} \left[x^* + \frac{(1+\gamma)x^{*2}}{2L} \right] \quad (17)$$

$$f(x^*, \dot{x}^*) = - \left[f_c(x^*, \dot{x}^*) + p_0 \pi \frac{D^2}{4} + \left(\frac{p_w S_2}{2} + F_r \right) \text{sign}(\dot{x}^*) + p_w \left(S_1 - \frac{S_2}{2} \right) - p_c S_3 \right] \quad (18)$$

The first order approximate solution of Eq. (16) and the approximate solution is given by

$$\begin{cases} x^* = a \cos \varphi + b + \frac{1}{m} x_1^*(a) \\ \dot{x}^* = \frac{1}{m} A_1(a) \\ \dot{\varphi} = \Phi_0(a, \varphi) + \frac{1}{m} \Phi_1(a, \varphi) \end{cases} \quad (19)$$

where a is amplitude, b is decentration, φ is phase, x_1^* and A_1 are functions of the amplitude, Φ_0 and Φ_1 are functions of the amplitude and the phase.

5. Results and discussion

5.1. Model validation

The measurement system of the prototype had pressure measurements on the combustion cylinder, the hydraulic chambers and in the common pressure rails, as shown in Fig. 6. The used measuring sensors are presented in Table 1. The piston displacement has been measured directly by a laser displacement sensor, since there is no mechanisms coupled to the SPHFPE piston assemble. The measured data synchronisation of the combustion cylinder pressure and the piston displacement is important and the data acquisition system PXI-6123 has been used. Further, the maximum piston acceleration appears at the same time with the highest com-

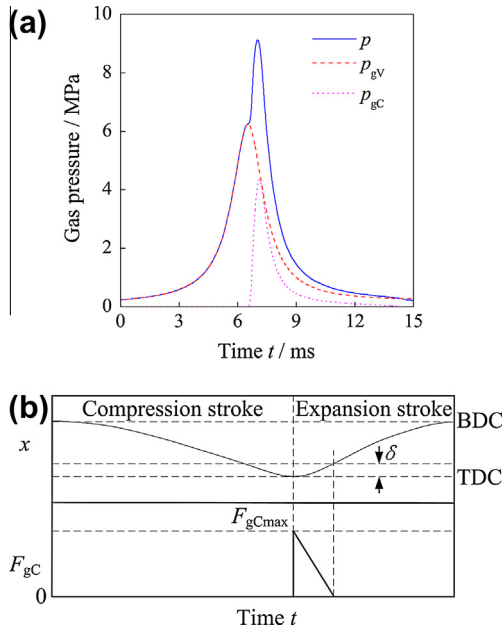


Fig. 5. The simplified combustion gas force. (a) Gas pressure components. (b) Simplified method.

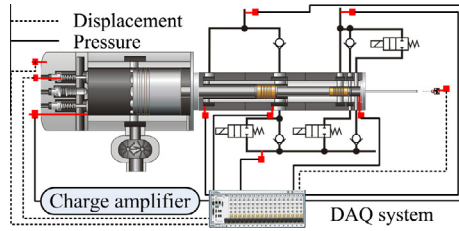


Fig. 6. Schematic diagram of experimental test rig.

Table 1
Used measuring sensors.

Measuring parameter	Sensors
Combustion cylinder pressure	Kistler 6055BB, Charge Amplifier 5011B
Hydraulic chamber pressures	HYDAC HDA-3844
Piston displacement	Laser displacement sensor ZLDS100/HS
Exhaust valve stroke	Laser displacement sensor LD1607
Engine body axis displacement	LVDT-EK1000

bustion cylinder pressure should be mentioned. The information was collected by National Instruments data acquisition system PXI-8196 and PXI-6123. The response of the data acquisition system is 50 kHz and the frequency has been applied to all the measured parameters individually. Further, a DAQ-board 6259 from National Instruments is also used to collect both the control signals and the sensor information. The response of the data acquisition system is 5 kHz individually.

The operation parameter of the SPHFPE prototype is presented in Table 2. The calculated piston oscillation was compared to data from the SPHFPE prototype, as shown in Fig. 7. The comparison was done with the aim of verifying that the model produces realistic results, and that it is able to predict real trends for varying engine design parameters.

5.2. Transient process

Before the SPHFPE piston oscillation takes place, initial oscillation energy is needed. The initial oscillation energy provides the gas compression energy of the start engine cycle and is stored in the compression accumulator. Without the initial oscillation energy, the SPHFPE piston oscillation never starts by itself. It seems that the engine piston dynamic system is a hard self-excited oscillation system.

The phase locus of different start compression positions is shown in Fig. 8. The piston oscillation is an unsymmetrical oscillation. It seems that the start oscillation position affects the phase locus of the engine little. However, the compression ratio of the start engine cycle should be higher enough since the SPHFPE is a diesel engine. The initial oscillation energy stored in the compression accumulator is transformed into the gas internal energy in the start engine cycle. It is seen that the piston oscillation becomes stable with the constant fuel combustion energy and the constant hydraulic working pressure. For the SPHFPE, the energy released by the fuel combustion is important for the piston oscillation sta-

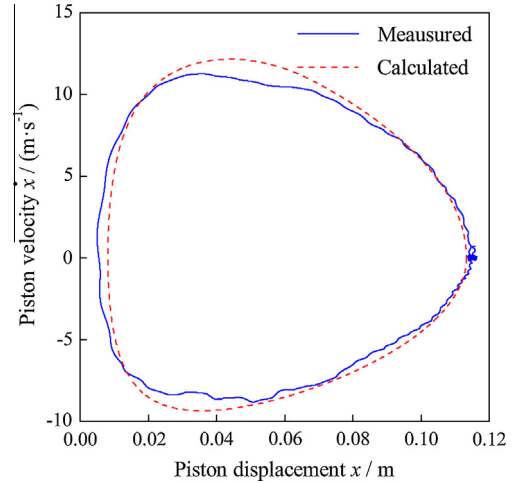


Fig. 7. Measured and calculated piston motion under given parameters.

bility. The released energy determines the expansion stroke length since the hydraulic working pressure changes little. The variable expansion stroke changes the BDC. The flow area of the intake port is affected by the BDC changes. Thus, the scavenging effect is mostly determined by the BDC under the constant scavenging gas pressure. The released energy needs to make the piston move back to the BDC. Then, the intake port is opened widely enough. The piston oscillation limit cycle is determined by the energy balance between the fuel combustion energy and the hydraulic load energy. It seems that a limit cycle appears in any piston stroke if the stable fuel combustion energy is supplied.

In the steady operation, the piston oscillation is a periodic oscillation. An equivalence damping can be used instead of the generalised damping in the steady operation. The calculation of the equivalence damping is based on the equality of the energy consumed by the generalised damping. The equivalence damping is calculated by

$$\int_{x_{CSBDC}}^{x_{ESBDC}} c(\dot{x}) dx = \int_{x_{CSBDC}}^{x_{ESBDC}} c_{equi}(\dot{x}) dx \quad (20)$$

where x_{CSBDC} is BDC in the compression stroke, x_{ESBDC} is BDC in the expansion stroke, $c_{equi}(\dot{x})$ is equivalence damping.

Then, Eq. (4) can be transformed to

$$m\ddot{x} + K(x, \dot{x})x = 0 \quad (21)$$

Base on Eq. (21), an amplitude frequency response characteristic of the equivalence oscillation system is given. The compression ratio ε which is especially important for the compression ignition engine is confirmed. The relationship between the compression ratio and the piston oscillation frequency f is presented based on Eq. (21), as shown in Fig. 9. The designed piston moving mass of the SPHFPE is about 4.2 kg. It is seen that the piston oscillation frequency has to satisfy the minimum compression ratio requirement for the compression ignition. The compression ratio increases with the increase of the piston oscillation frequency. Because the output power is in direct proportion to the piston oscillation frequency, an

Table 2
Used control parameters.

Parameter	Value	Parameter	Value
Output pressure	12.8 MPa	Low-pressure rail	1.2 MPa
Compression pressure	14.0 MPa	Injector driving pressure	14.0 MPa
Injection timing (before TDC)	23 mm	Valve timing (after TDC)	58 mm
Exhaust valve driving pressure	14.0 MPa	Exhaust valve signal pulse width	10 ms

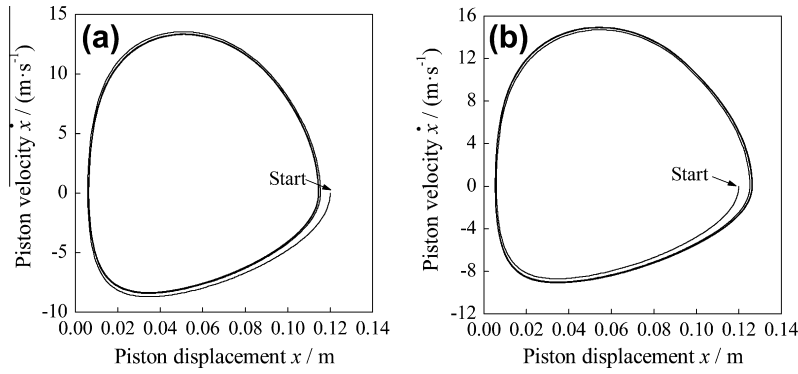


Fig. 8. The phase locus under different piston strokes. (a) A phase locus with a decreasing piston stroke. (b) A phase locus with an increasing piston stroke.

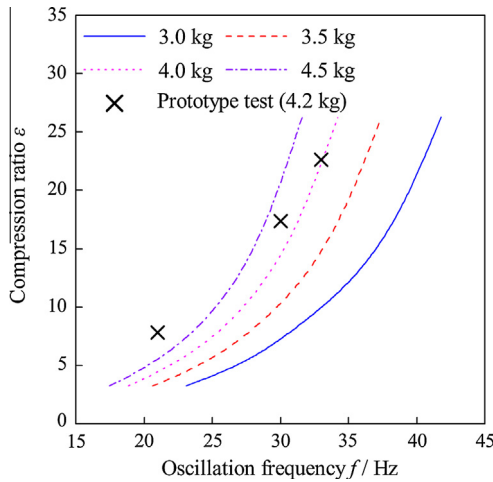


Fig. 9. Compression ratio vs. oscillation frequency of different piston masses.

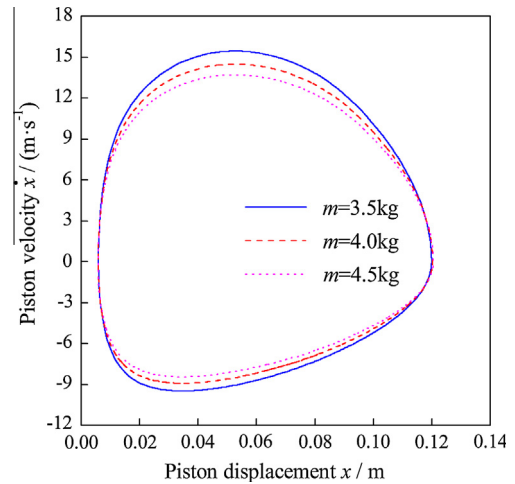


Fig. 10. The phase locus of different piston moving masses.

effective method is required to change the output power of the SPHFPE. To get a lower piston oscillation frequency, a pulse pause modulation control method has been proposed [4]. The pulse pause modulation control does not change the piston oscillation characteristics of one cycle. Thus, the compression ratio requirement and the controllable output power are satisfied at the same time.

5.3. Parameter sensitivity

The piston oscillation characteristics change with different engine design parameters. To indicate the changing characteristics of the piston motion more clearly, the relation between the piston displacements x and x^* is given by

$$\begin{cases} x = L - x^* \\ \dot{x} = -\dot{x}^* \end{cases} \quad (22)$$

Fig. 10 shows the piston oscillation trajectory at the different piston moving mass. It seems that the change of the piston moving mass have little effect on the engine dead centre. The different piston moving mass only leads to changes in both the piston velocity and the piston oscillation frequency. The engine gets a higher piston oscillation frequency with a smaller piston moving mass. However, the higher piston oscillation frequency demands a much faster response of the engine control system. The exhaust valve needs a fast control unit, which has been designed for the SPHFPE

[25]. The spool of the check valve on the pump chamber also need to move fast to reach a higher volumetric efficiency. Some methods have been proposed to satisfy the fast response requirement.

Fig. 11 proposes the piston oscillation trajectory and the dead centre change at the different hydraulic working pressure. The change rate of a design parameter is calculated by

$$\omega = 100 \times \frac{Para - Para_{rated}}{Para_{rated}} \quad (23)$$

where $Para$ is design parameter value, $Para_{rated}$ is design parameter rated value, ω is change rate. It is seen that the change of the hydraulic working pressure has a direct effect on the piston stroke because of the energy balance in the engine operation. Both TDC and BDC have shown a high sensitivity to the hydraulic working pressure. With the increase of the hydraulic working pressure, the TDC position increases and the compression ratio becomes smaller. The change rate of the compression ratio is about two times larger than the hydraulic working pressure change rate. The increase of the hydraulic working pressure may lead to misfire since the increase of the hydraulic working pressure has a great effect on the decrease of the compression ratio. The increase of the hydraulic working pressure also causes the reduction of the BDC. The piston stroke also decreases. It seems that a stable hydraulic working pressure is favourable for the engine operation. The minimum compression ratio of the engine should be guaranteed in the engine design when the maximum hydraulic working pressure is applied. Further, the measured hydraulic working pressure needs to be set as a load

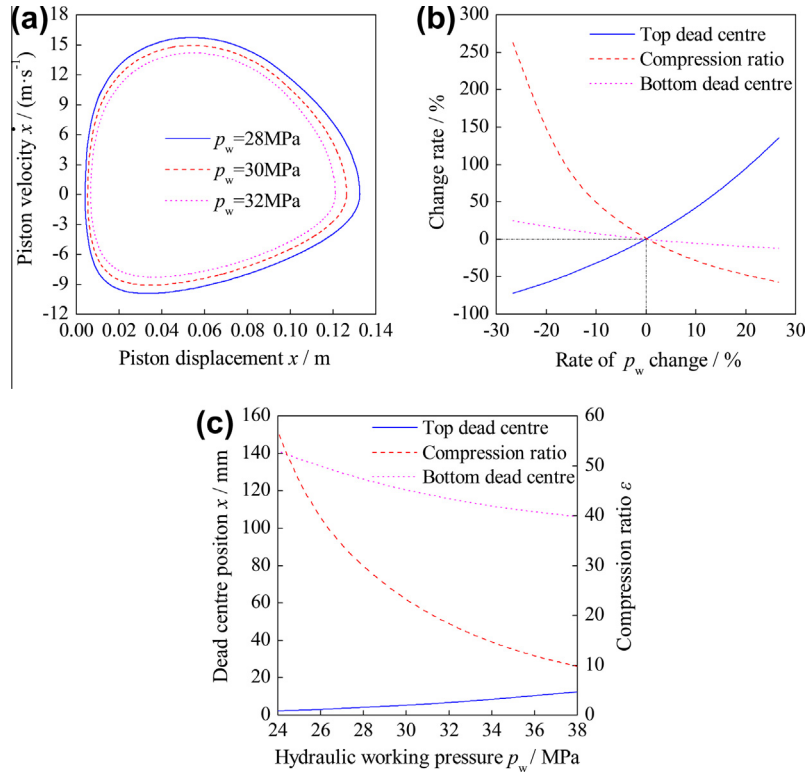


Fig. 11. The phase locus of different hydraulic working pressures. (a) Phase locus variety. (b) Dead centre changes. (c) Values of the dead centres.

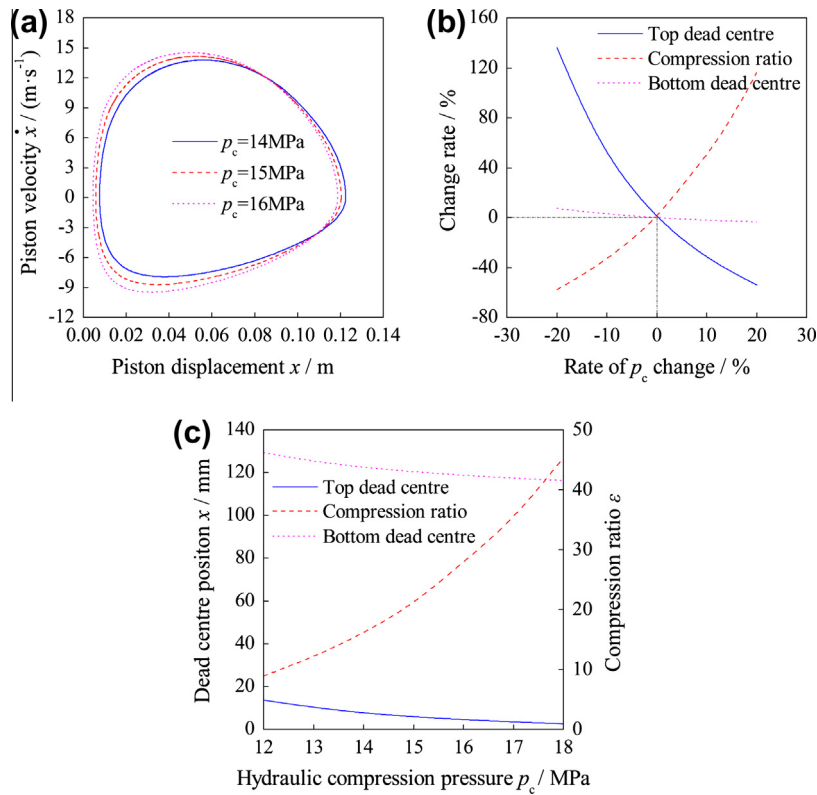


Fig. 12. The phase locus of different hydraulic compression pressures. (a) Phase locus variety. (b) Dead centre changes. (c) Values of the dead centres.

feedback for the engine control unit design, since the load of the engine has a linear relationship with the pressure.

Fig. 12 shows the piston oscillation trajectory of different hydraulic compression pressures. It can be seen that the hydraulic

compression pressure has a much larger effect on the TDC than the BDC. The piston is free to move linearly and the compression stroke is realised by the hydraulic energy provided by the hydraulic accumulator. The hydraulic compression pressure in the hydraulic

accumulator is an important parameter for the compression ratio control. An approximately linear relationship between the hydraulic compression pressure and the TDC is shown. Thus, the hydraulic compression pressure largely determines the compression ratio. The compression ratio can be adjusted freely by controlling the hydraulic compression pressure. The engine can also get a better compression ignition performance by increasing the hydraulic compression pressure, especially in the start engine cycle. Further, the piston stroke length changes little when the hydraulic compression pressure changes. To get an accurate compression ratio control, the changing of the hydraulic compression pressure needs an additional hydraulic control system. However, the minimum compression ratio of the engine should be achieved in the compression stroke with the minimum hydraulic compression pressure. Achten and co-researchers [4] have proposed a method to change the hydraulic compression pressure in the compression accumulator by means of electrohydraulic valves. The hydraulic compression pressure represents the compression energy, because the gas precharge pressure of the compression accumulator is constant. The amount of compression energy and subsequently the compression ratio can be controlled by the method. The changes of the hydraulic working pressure may change the BDC of the next cycle. It means the compression energy changes if the hydraulic compression pressure is constant. The compression ratio becomes variable, which is unfavourable for the engine stable operation. For the variations in the hydraulic working pressure, the hydraulic compression pressure of the next cycle should have the same change trend with the hydraulic working pressure. It is aimed to achieve a stable compression ratio under a variable piston stroke.

Fig. 13 presents the piston oscillation trajectory changes at the different fuel combustion energy. It can be seen that the expansion stroke length is deeply affected by the fuel combustion energy. The stroke length is a result of the energy balance between the fuel combustion energy and the hydraulic load energy in the expansion stroke. The smaller fuel combustion energy causes a smaller BDC.

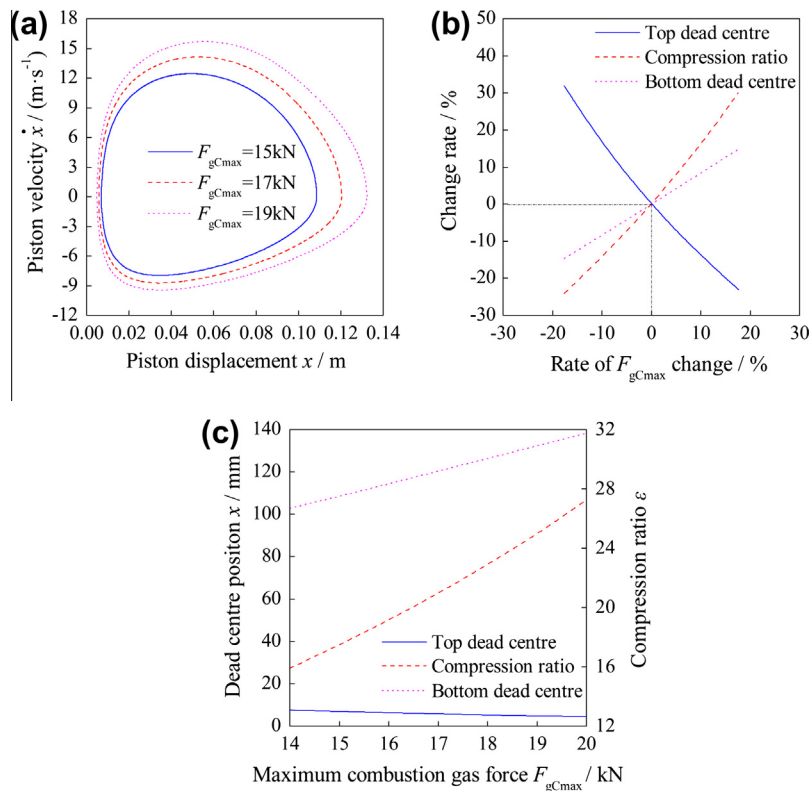


Fig. 13. The phase locus of different combustion energies. (a) Phase locus variety. (b) Dead centre changes. (c) Values of the dead centres.

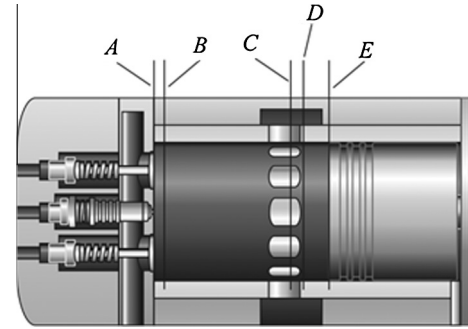


Fig. 14. Special piston positions. A is the position of the cylinder head plate. B is determined by the minimum compression ratio. C is determined by the smallest flow area of the scavenging port. The length between C and D is determined by the piston rebound. E is determined by the hydraulic pump end closure.

The TDC becomes larger and the compression ratio becomes smaller. The trend is bad for the stability of the engine operation. The accurate description of the combustion heat releasing is still a big challenge in the free-piston engine researches. Further, the combustion heat releasing is also rapid and the short combustion duration decreases the rated engine power [24]. The smaller rated engine power has a negative effect on the FPE application field development. Higher fuel mass flow for a faster combustion of one cycle is needed to increase the rated engine power. In the SPHFPE control system, a fuel injection control strategy is used with the piston displacement feedback. When the BDC becomes too small, a little larger fuel mass is injected. The control is to make the piston go back to the rated BDC reliably.

5.4. Limit cycle stability

The piston oscillation limit cycle appears when there is sufficient combustion energy released. To realise the fuel combustion,

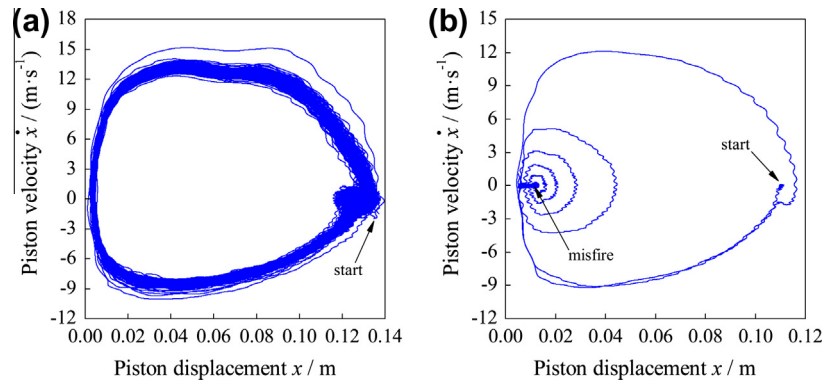


Fig. 15. Measured piston oscillation characteristics. (a) Stable piston oscillation. (b) Unstable piston oscillation and misfire.

an enough compression ratio is important for a compression ignition engine. The compression ratio requirement for the SPHFPE compression ignition is not satisfied in most small piston displacements. Too small piston displacement also affects the SPHFPE operation, especially the scavenging effects. Guaranteeing the scavenging effect can improve the engine operation performance. To achieve a better scavenging effect, the sufficient flow area of the scavenging port is required. Thus, the minimum BDC should be confirmed, as shown in Figs. 14 and 15. The actual BDC should be controlled to be big enough to ensure suitable flow areas of the scavenging ports. Further, the scavenging effect is determined not only by the flow area of the scavenging port, but also by the scavenging pressure. The higher scavenging pressure increases the scavenging performance. However, the higher hydraulic compression pressure is also needed to achieve the same compression ratio under the higher scavenging pressure. For the SPHFPE, the maximum BDC is restricted by the hydraulic pump end closure. Further, if the piston moves far outside the rated BDC, it leads to a mechanical contact between the piston and the end closure head. The mechanical contact damages the mechanical structure of the SPHFPE.

In the practical application, the SPHFPE often works under its self-exacted frequency since the output power is not always the largest. Therefore, the piston oscillation frequency needs to be controlled. The control is realised by stopping the piston around BDC for a moment. The piston oscillation characteristics will not be changed under the control, because the piston oscillation is a hard self-oscillation. The control parameters are included in the generalised damping and the generalised nonlinear stiffness. The scavenging pressure, the injection timing, the injected fuel mass, the hydraulic working pressure and the hydraulic compression pressure are included in the control parameters. The oscillation results propose the right direction to the appropriate parameter set of the SPHFPE control system as soon as the operation condition of the engine changes. Other contributing factors such as the hydraulic leakage and the hydraulic valve dynamic should also be considered in the engine operation. The effects of these factors could be achieved by both the full-cycle model simulation and the prototype test.

6. Conclusions and future work

The semi-analytical method for the SPHFPE parameter design was proposed. The results indicate that the piston oscillation has an unsymmetrical oscillation characteristic. The oscillation is a hard self-oscillation with variable damping and stiffness. The external energy is required to start the piston oscillation. The stability of the piston oscillation is determined by the energy balance between the fuel combustion energy and the hydraulic load. The

compression ratio increases with the higher piston oscillation frequency. The piston oscillation frequency should satisfy the minimum compression ratio requirement for the compression ignition. In the steady operation, the dead centres of stable piston trajectories are mainly determined by the hydraulic working pressure, the hydraulic compression pressure and the fuel combustion. With the increase of the hydraulic working pressure, the piston stroke decreases and the compression ratio becomes smaller. An approximately linear relationship appears between the hydraulic compression pressure and the TDC. The expansion stroke is deeply affected by the fuel combustion energy. The scavenging affected by the BDC should be paid special attention to ensure the stable piston oscillation.

The results show the right direction to the appropriate parameter set of the SPHFPE control system as soon as the operation condition changes. High reliability and fast response of engine control system are significant for the SPHFPE piston motion control. Much development work remains before the SPHFPE could be used for the realistic automotive application. Work on these topics is currently underway in the National Key Laboratory of Vehicular Transmission at the Beijing Institute of Technology.

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References

- [1] Pescara RP. Motor compressor apparatus. US Patent 1657641; 1928.
- [2] Achten PAJ. A review of free piston engine concepts. SAE paper 941776; 1994.
- [3] Mikalsen R, Roskilly AP. A review of free-piston engine history and applications. *Appl Therm Eng* 2007;27:2339–52.
- [4] Achten PAJ, Van Den Oever JPJ, Potma J, Vael GEM. Horsepower with brains: the design of the Chiron free piston engine. SAE paper 2000-01-2545; 2000.
- [5] Hibi A, Ito T. Fundamental test results of a hydraulic free piston internal combustion engine. *Proc Inst Mech Eng Part D J Automob Eng* 2004;218:1149–57.
- [6] Mikalsen R, Roskilly AP. The design and simulation of a two-stroke free-piston compression ignition engine for electrical power generation. *Appl Therm Eng* 2008;28:589–600.
- [7] Xu SQ, Wang Y, Zhu T, Xu T, Tao CJ. Numerical analysis of two-stroke free piston engine operating on HCCI combustion. *Appl Energy* 2011;88:3712–25.
- [8] Underwood AF. The GMR 4–4 “HYPREX” engine a concept of the free-piston engine for automotive use. SAE paper 570032; 1957.
- [9] Amann CA. Evaluating alternative internal combustion engines: 1950–1975. *J Eng Gas Turb Power Trans ASME* 1999;121:540–5.
- [10] Frey DN, Klotzsch P, Egli A. The automotive free-piston-turbine engine. SAE paper 570051; 1957.
- [11] Zhao ZF, Zhang FJ, Huang Y, Zhao CL, Guo F. An experimental study of the hydraulic free piston engine. *Appl Energy* 2012;99:226–33.
- [12] Jakob F, Ingemar D. Simulation of a two-stroke free piston engine. SAE paper 2004-01-1871; 2004.

- [13] Mikalsen R, Roskilly AP. Performance simulation of a spark ignited free-piston engine generator. *Appl Therm Eng* 2008;28:1726–33.
- [14] Hu JB, Wu W, Yuan SH, Jing CB. Mathematical modelling of a hydraulic free-piston engine considering hydraulic valve dynamics. *Energy* 2011;36:6234–42.
- [15] Xiao J, Li QF, Huang Z. Motion characteristic of a free piston linear engine. *Appl Energy* 2009;87:1288–94.
- [16] Li QF, Xiao J, Huang Z. Simulation of a two-stroke free-piston engine for electrical power generation. *Energy Fuel* 2008;22:3443–9.
- [17] Jaakko L, Juha H, Petri S, Jari B. Fluid dynamic modelling of a free piston engine with labyrinth seals. *J Therm Sci* 2010;19:141–7.
- [18] Wu W, Yuan SH, Hu JB, Jing CB. Design approach for single piston hydraulic free piston diesel engines. *Front Mech Eng China* 2009;1:371–8.
- [19] Chiang CJ, Yang JL, Lan SY, Shei TW, Chiang WS, Chen BL. Dynamic modeling of a SI/HCCI free-piston engine generator with electric mechanical valves. *Appl Energy* 2013;102:336–46.
- [20] Mikalsen R, Roskilly AP. Coupled dynamic-multidimensional modelling of free-piston engine combustion. *Appl Energy* 2009;86:89–95.
- [21] Mao JL, Zuo ZX, Feng HH. Parameters coupling designation of diesel free-piston linear alternator. *Appl Energy* 2011;88:4577–89.
- [22] Mao JL, Zuo ZX, Li W, Feng HH. Multi-dimensional scavenging analysis of a free-piston linear alternator based on numerical simulation. *Appl Energy* 2011;88:1140–52.
- [23] Formosa F. Coupled thermodynamic–dynamic semi-analytical model of free piston stirling engines. *Energy Convers Manage* 2011;52:2098–109.
- [24] Hu JB, Wu W, Yuan SH, Jing CB. Fuel combustion under asymmetric piston motion: tested results. *Energy* 2013;55:209–15.
- [25] Huang Y, Guo F, Zhao CL, Zhang FJ. Study on electrohydraulic exhaust valve of single piston hydraulic free-piston engine. *Adv Sci Lett* 2012;6:436–40.